

A New Analysis Approach for Motorcycle Brake Squeal Noise and Its Adaptation

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ABSTRACT

Concerning the potential for low frequency brake squeal noises from motorcycles, the issue of the noise caused by the structural instability of the system was addressed with mode-coupling mechanism analysis utilizing complex eigenvalue analysis. The equation of motion of the brake system was simplified by converting it from the physical domain to the modal domain. The parameters required for the stability analysis were obtained from the normal mode solution of the system model. Moreover, it was possible to decrease the time required for repeat calculations with different friction coefficients and to complete instability-cause analysis in a mode by 70% or more by using a numerical-analysis solver. A highly accurate analysis model was established and applied in the new approach to specify the modes affecting squeal noises and to study countermeasures. From the test results for the subsequently modified brake system, it was verified that the new method is effective for quickly predicting and making countermeasures for brake squeal noises in the design stage.

1. Introduction

Brake noise adversely affects customer satisfaction, and, further, a number of man-hours are required to resolve the problem of brake noise. Thus, it is an important technical issue in the development of motorcycles. Squeal noise is one of the harshest of all brake noises to the ear. It is hypothesized that there are two generating mechanisms for squeal noise. The first mechanism is a phenomenon resulting from “stick-slip” on the friction side. The second mechanism is a phenomenon resulting from geometric instabilities of the brake assembly. One recent approach to the squeal noise caused by geometric instability of the system is the application of mode-coupling analysis of the mechanism using complex eigenvalues analysis. However, in the current complex eigenvalue method, complex eigenvalues are merely plotted on a complex plane. Thus,

when the density of the mode is high, the identification of unstable modes is very difficult. Furthermore, complex eigenvalues are generally solved using MSC-NASTRAN. Since system model computation is only possible with one condition at a time, much time is needed in order to obtain a root locus by numerous calculations with different coefficients of friction. Therefore, phenomenon analysis and design change analysis are difficult. The authors have developed a new modeling and numerical analytical method, which makes it possible to obtain a root locus in a short time. At the same time, a new evaluation scale was developed to express the coupling strength of an unstably coupled mode⁽¹⁾.

Generally, squeal noise is defined to be in the range from 1 kHz to about 15 kHz. The range is divided into a low frequency domain (1 to 3 kHz) and a high frequency domain (5 kHz and above). In this research, the new

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analytical method concerning the geometric instability of the brake system was applied to the low-frequency squeal noise (1 kHz level) of motorcycle brakes to prove the appropriateness of the method.

2. New Analysis Approach

2.1. Simplification of the Equation of System Motion

The equation of system motion was greatly simplified by shifting the analysis of stability from physical to modal domains. The general equation of system motion can be expressed, in matrix form, as shown in equation (1).

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F_f\} \quad (1)$$

where, M , C , and K are the mass, viscous damping, and stiffness matrices respectively for the non-friction system, u is the displacement vector of the system, F_f is the friction force applied to the system. A dot over the displacement denotes differentiation with respect to time. The stability evaluation model represents the contact surface between the disc and pads in the direction normal to the surface only. The friction force is derived by the following equation.

$$F_f = \mu N \quad (2)$$

N is the force between the pad and disc normal to the contact surface. μ is the coefficient of friction. Brake system instabilities can result if either the coefficient of friction or the normal force vary. The theory of brake squeal stability used here is such that the coefficient of friction is constant and the normal force varies as a function of the vibration of the pad and disc. The friction equation is then modified as follows.

$$\{F_f\} = \mu(\{N_{static}\} + \{N_{dynamic}\}) \quad (3)$$

While solving for eigenvalue, the static term will be dropped from the dynamic equations. The dynamic normal force is caused by the vibration of the pad and disc. The contact points are connected with a simple spring K_s . Dynamic friction force is modeled by the following equation.

$$\{N_{dynamic}\} = K_s (\{u_{N,disc}\} - \{u_{N,pad}\}) \quad (4)$$

where $u_{N,pad}$ denotes the displacement of the pad in the normal direction. The resulting friction force, F_f , is applied to degree of freedom (DOF) $u_{T,pad}$ and $-F_f$ is applied to DOF $u_{T,disc}$, where T denotes the tangential direction.

The normal force is determined by the relative displacement (*rel*) between node pairs as using equation (5).

$$F_{T,rel} = -\mu \times K_s \times u_{N,rel} \quad (5)$$

Based on equations (2) – (5), the equation of motion (1) becomes

$$[M_g]\{\ddot{v}\} + [C_g]\{\dot{v}\} + [K_g]\{v\} = -\mu[K_f]\{v\} \quad (6)$$

where M_g , C_g and K_g are mass, damping and stiffness matrices, which includes the generalized DOF. The vector of the DOF is then as indicated in equation (7).

$$\{v\} = \begin{Bmatrix} u \\ \vdots \\ u_{N,rel} \\ u_{T,rel} \end{Bmatrix} \quad (7)$$

and the friction force term is as indicated in equation (8).

$$[K_f] = \begin{bmatrix} [0] & [0] & [0] \\ [0] & [0] & [0] \\ [0] & [K_{TN}] & [0] \end{bmatrix} \quad (8)$$

By using Laplace conversion on equation (6), we arrive at system equation (9). For the stability analysis, we omit the viscous damping term $[C_g]$ from the system equation of motion, the purpose being to simplify the equation. The damping curve will be defined in the final stability analysis.

$$s^2[M]\{\bar{v}\} + [K]\{\bar{v}\} = -\mu[K_f]\{\bar{v}\} \quad (9)$$

where $\{\bar{v}\} = Laplace\{v\}$.

In order to convert equation (9) to a modal domain, equation (10) is utilized.

$$\{\bar{v}\} = [\psi]\{\bar{\gamma}\} \quad (10)$$

where $[\psi]$ is the mode shape and $\{\bar{\gamma}\}$ is the modal displacement.

By substituting equation (10) into a pre-multiplied equation (9) by $[\psi]^T$ the final system equation of motion in the modal domain is obtained (equation 11).

$$\left(s^2 + \begin{bmatrix} \omega_1^2 & 0 & \cdots & 0 \\ 0 & \omega_2^2 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & \omega_n^2 \end{bmatrix} + \mu[\Lambda_f] \right) \{\bar{\gamma}\} = \{0\} \quad (11)$$

where the ω_i is the i -th natural frequency of the non-friction system.

From the definition of friction term $[\Lambda_f]$ in equation (8),

$$[\Lambda_f] = [\psi]^T [K_f] [\psi] = K_s \cdot f([\psi_{T,rel}], [\psi_{N,rel}]) \quad (12)$$

$\psi_{T,rel}$ is the mode shape of contact point in the tangential direction.

$\psi_{N,rel}$ is the mode shape of contact point in the normal direction.

All parameters of the equation of motion (11) can be obtained from the normal mode analysis of a non-friction system. The necessary parameters are natural frequencies and the mode shapes of the contact points. All parameters represent real values and the results are all highly accurate. The complex eigenvalue in equation (11) can then be computed in a numerical analysis solver such as Matlab of The MathWorks Inc. Even when evaluating the complex eigenvalues with different coefficients of friction, there is

no need to change the model set up, and thus the results can be obtained immediately. It is possible to significantly decrease the amount of time required for computations by 70% compared to the conventional method.

2.2. New Evaluation Scale for the Instability Mode

Complex eigenvalue analysis of the actual part size can provide us with information about the probability of a squeal noise occurring. However, it does not provide us with any information regarding the magnitude of that noise.

Further, model accuracy, connecting conditions, boundary conditions, and natural frequency variations due to the tolerance of components in the analytic model are sometimes larger than the gap between the two modes, and may significantly affect the prediction of the complex eigenvalues. Thus, it is necessary to have evaluation criteria to determine the degree that the system is latently unstable as a portion of the squeal stability evaluation method. When the coupling strength is strong, the brake system possesses characteristics which can lead to squeal. The physical meaning of coupling strength is how quickly the eigenvalues converge when the coefficients of friction increase from zero.

Therefore, we define the new Evaluation Scale for the Instability Mode. It expresses qualitatively the coupling strength of the coupling mode.

The coupling strength between the two unstable modes which converge and split when the coefficient of friction increases can be expressed by CS_{il} in equation (13)⁽¹⁾.

$$CS_{il} = \frac{[\Lambda_f]_{il} \cdot [\Lambda_f]_{li}}{(\omega_i^2 - \omega_l^2)} \quad (13)$$

We use the complex eigenvalue and the coupling strength (CS) to evaluate the probability of brake squeal.

3. Examples of Brake Squeal Analysis in Motorcycles

3.1. Construction of Motorcycle Brake Systems

The brake system for motorcycles as used in the present study consists of twin piston floating-type calipers as shown in Fig. 1. The calipers are fastened to the front fork by bolts through the support. The brake discs are fastened to the wheel by bolts.



Fig. 1 Front brake system of test motorcycle

3.2. Understanding the Squeal Noise Phenomenon

Brake squeal noise was verified with a brake noise tester using an actual motorcycle. The characteristics of squeal noise were judged in a matrix mode simulating the conditions of users applying the brake with speed, extent of deceleration, temperature and applied pressure as parameters. The test results reveal that squeal noise is generated in the low deceleration zone of about 1.2 kHz as shown in Fig. 2.

3.3. Preparation of Analysis Model and Correlation

3.3.1. Component models for analysis

FE models were made for the disc, calipers, friction pads, support and the wheel. In order to ultimately obtain a reliable brake-system model, a high correlation between the FEM frequency-response function (FRF) and the impact excitation test are required for every component. Therefore, the model geometry and the physical property values are adjusted to exact values. The measurement points on the disc are shown in Fig. 3. The correlation of the impact excitation test and the FEM results is shown in Fig. 4.

The correlation was similar for the component parts. Another example is provided below. The measurement points on the caliper are shown in Fig. 5. The correlation of the impact excitation test and the FEM results is shown in Fig. 6.

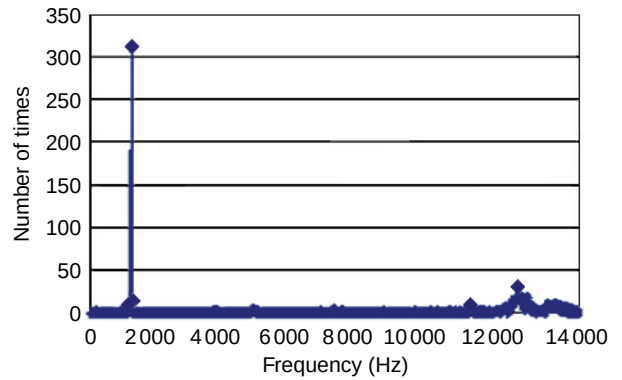


Fig. 2 Histogram of squeal noises

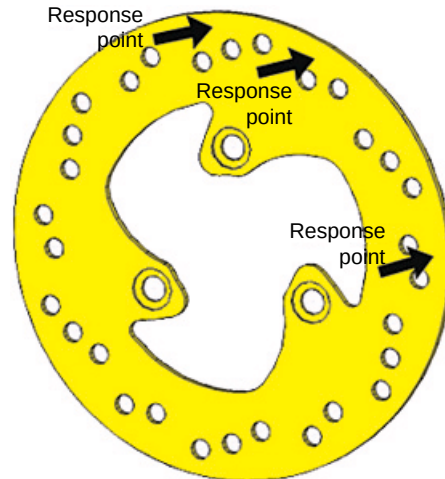


Fig. 3 FRF measurement points

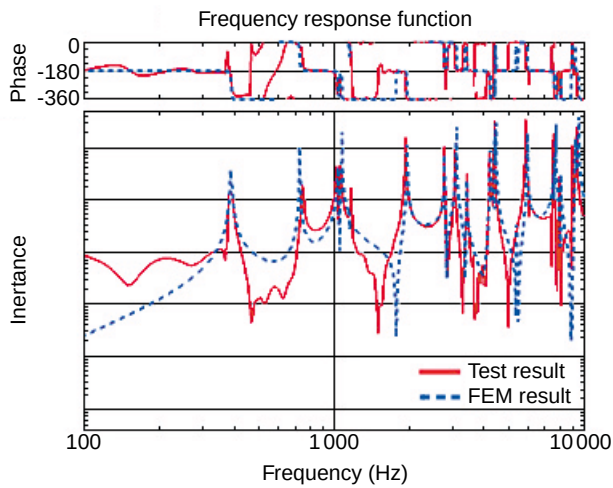


Fig. 4 Correlation sample of disc

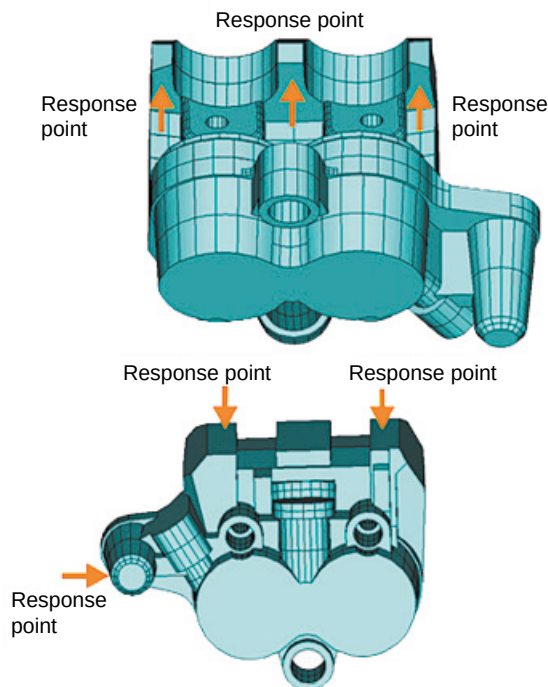


Fig. 5 FRF measurement points

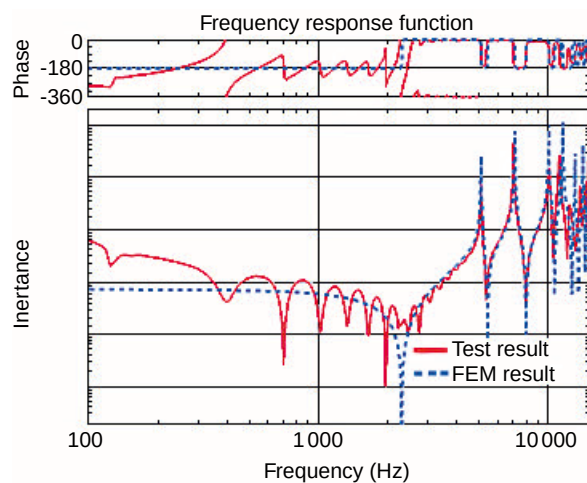


Fig. 6 Correlation sample of caliper

3.3.2. Sub-system construction and correlation

The connecting conditions of each part in the brake system defines the subsystem for every component, and the appropriate value was determined by verifying correlation.

(1) Modeling the brake disc and wheel subsystem

The brake disc is fastened to the wheel with bolts as shown in Fig. 7. The connecting conditions for the disc and wheel were determined using the correlation of the FRF in the target frequency domain. The correlation of the wheel and disc subsystem is shown in Fig. 8.

(2) Modeling the caliper and front fork subsystem

In the actual motorcycles, the calipers are fastened to the front fork with bolts through the support.

The connecting conditions between the support and the front fork, and the connecting conditions for the caliper and support were determined similarly.

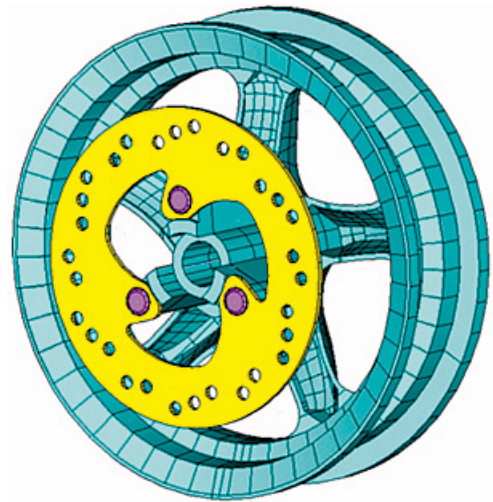


Fig. 7 Sub-system of wheel and brake disc

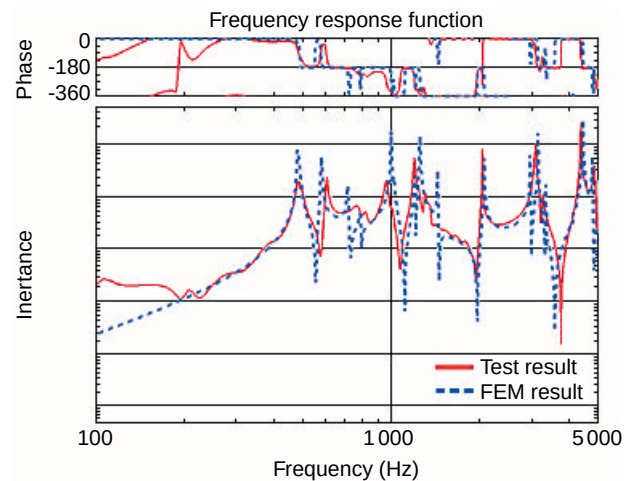


Fig. 8 Correlation of sub-system of wheel and brake disc

3.3.3. System model construction and correlation

The brake system model combined each subsystem, and the boundary conditions for the piston, the sliding section, the friction areas, etc. are also set to create a system model.

Correlation between the FRF calculation results for

the various boundary conditions are modified. In this way, a brake system model with the FRF characteristics of real motorcycle conditions was obtained. A system model is shown in Fig. 9. The correlation is shown in Fig. 10. There is correlation of the frequency domain in which squeal occurs between the characteristics of the analytic model and a real motorcycle. Thus, the validity of the boundary conditions was verified.

Differences in conditions under which squeal occurs can be responded to by adjusting the boundary conditions.

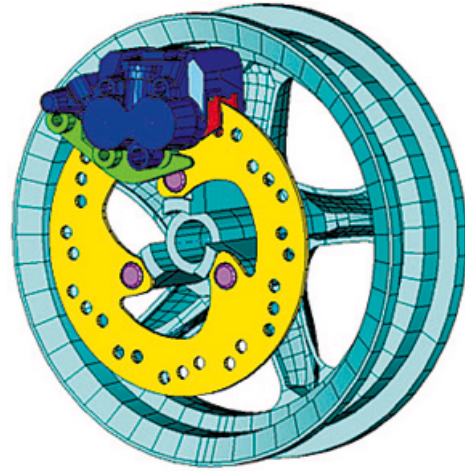


Fig. 9 Brake system model

4. Results and Mode Identification

4.1. Analysis Results

The parameters required for the equation of motion in the modal domain were determined by normal mode analysis of the system model. The root locus plots (Fig. 11, left) and coupling strength (Fig. 11, center) were computed by inputting them into the numerical value analysis solver together with different coefficients of friction.

This result expresses well the squeal noise phenomenon found near 1.2 kHz in actual real motorcycle measurements (Fig. 11, right). This indicates that this approach is effective.

4.2. Instability Mode Identification

The two coupling instability modes (1,220 Hz and 1,260 Hz) can be easily identified by the new stability metric (CS). The mode shape of the two instability modes was observed. The coupling pair mode shape is shown in Fig. 12.

The 1,220 Hz mode is the disc's '2.5 diameter node' mode. The 1,260 Hz mode is the wheel hub's torsion mode. The modes attributable to squeal were identified using this approach.

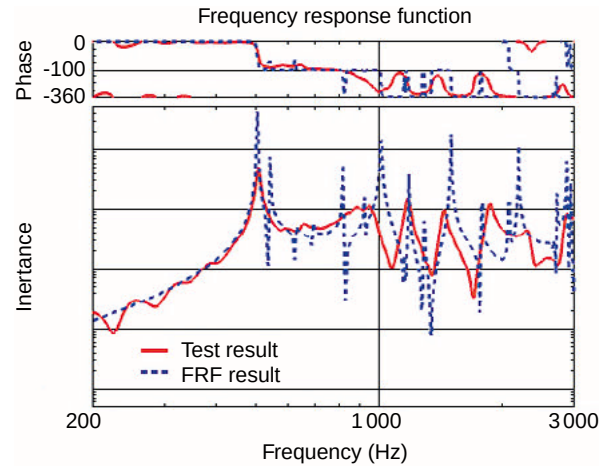


Fig. 10 Correlation in brake system model

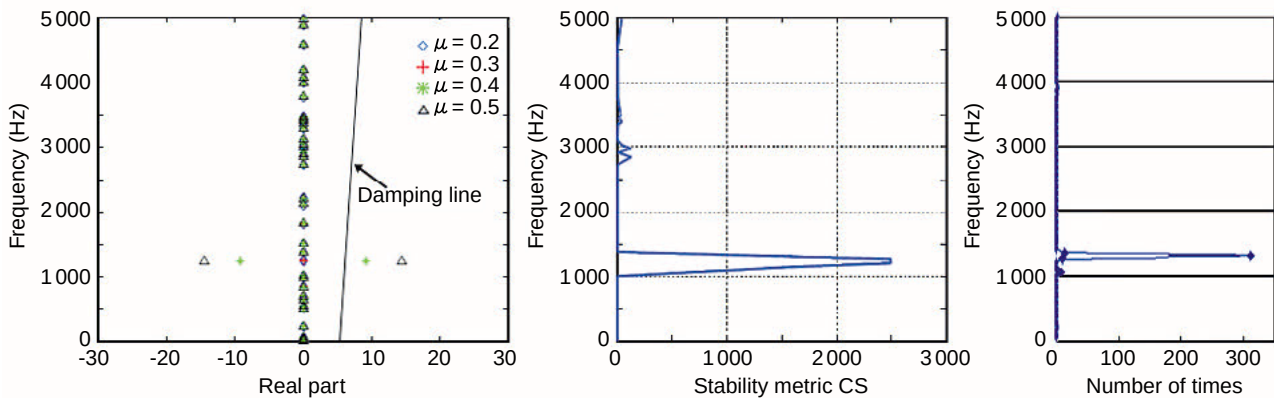


Fig. 11 Correlation between root locus plots, squeal noises and stability metric

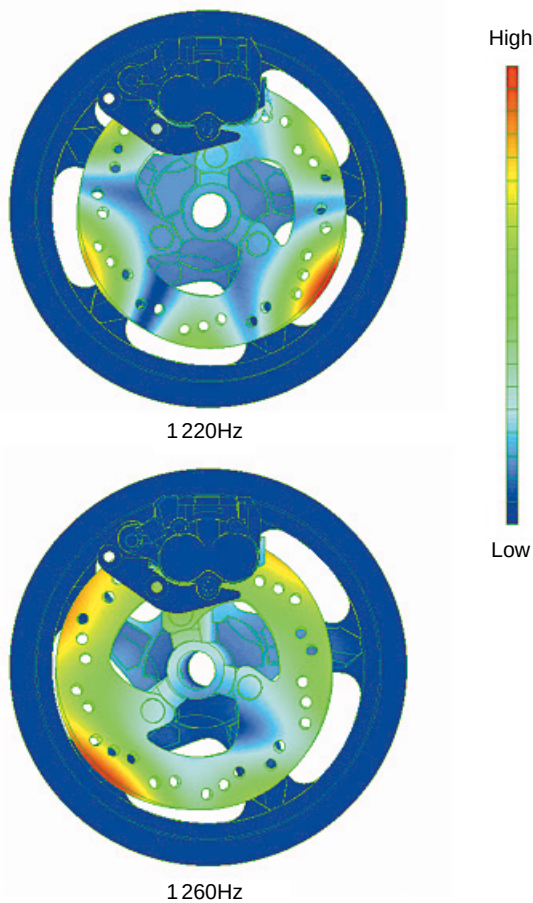


Fig. 12 Mode shape, pair modes

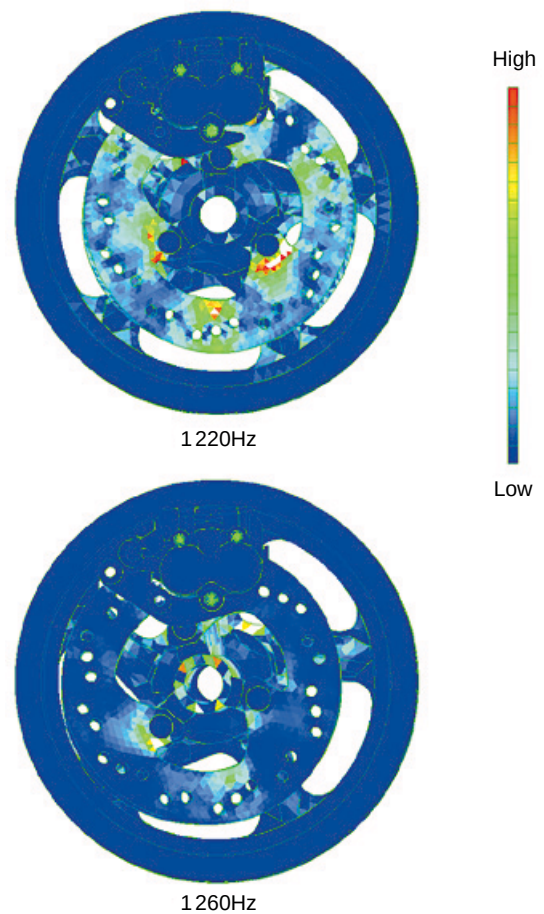


Fig. 13 Strain energy, pair modes

5. Examination of Countermeasure Specifications

5.1. Countermeasure Specification Concept

The strong coupling mode of the system converges and splits as the coefficients of friction increase. This indicates that there are two methods for addressing the issue of squeal noise. The first method is to separate the instability mode. That is, separating the eigenvalues of the two coupling modes so there is no high and positive damping even if affected by the coefficient of friction. The second method is to reduce the coupling strength of the instability mode. The examination of shape from these two viewpoints will provide countermeasures against brake squeal noises.

5.2. Examination of Countermeasure Specifications

This paper examined the separation in the two coupling pair mode. This approach is effective if attention is focused on the areas with high strain energy.

The strain energy of the coupling pair mode is shown in Fig. 13. The rate of distorted strain energy contribution for each part of the coupling pair mode is shown in Table 1. The 1,220 Hz mode has high strain energy in the disc attachment area. The 1,260 Hz mode has high strain energy

Table 1 Contribution rate of strain energy

	Mode number 14 Frequency 1 220(Hz)	Mode number 15 Frequency 1 260(Hz)
Parts group	Strain energy (%)	
Pad A	0.37	0.6
Pad B	0.5	0.52
Bolt	0.37	0.3
Support	0	0.04
Caliper	0.64	2.44
Backplate A	0.49	0.14
Backplate B	0.06	0.1
Disc	57.1	17.08
Spring Ks	10.7	34.95
Beam	0.03	0.12
Wheel	30.2	43.71

in the wheel hub. Here, we changed the shape of the disc attachment area and the mode eigenvalue mode. This enabled us to determine design specifications for the disc which will increase the connecting mode separation by 3.3%.

6. Evaluation of Design Change Specifications

6.1. Stability Analysis

The root locus plots and the coupling strength of the base specifications and the specifications of design change are shown in Fig. 14. The change in design specifications reduced positive damping and coupling strength. Brake specifications whereby squeal noises were not easily generated were predicted.

6.2. Prototype Evaluation

The predicted FRF and the FRF in the experiments are correlated, and it was verified that the characteristics of the design change specifications and the prototype corresponded, as shown in Fig. 15. It was confirmed that the prototype part was made according to the specifications.

6.3. Verification of Brake Squeal in Actual Motorcycles

The squeal noise was verified in a brake-noise tester using a motorcycle. The characteristics of squeal noise were judged in a matrix mode simulating the conditions of users applying the brake with speed, extent of deceleration, temperature and applied pressure as parameters.

The brake squeal occurrence frequency for the base specifications and the countermeasure specifications is shown in Fig. 16. Brake squeal disappeared in the vicinity of 1.2 kHz, our target problem area, confirming the effectiveness of the countermeasure. The charts on the left show the occurrence frequency of the noise. The charts on the right show the occurrence frequency for each condition.

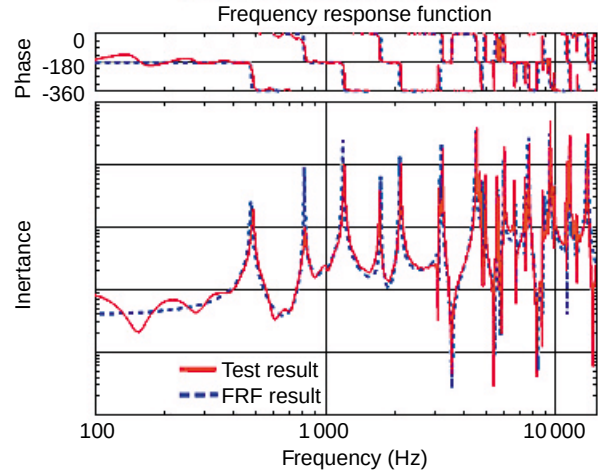


Fig. 15 Correlation between experiments and predicted values

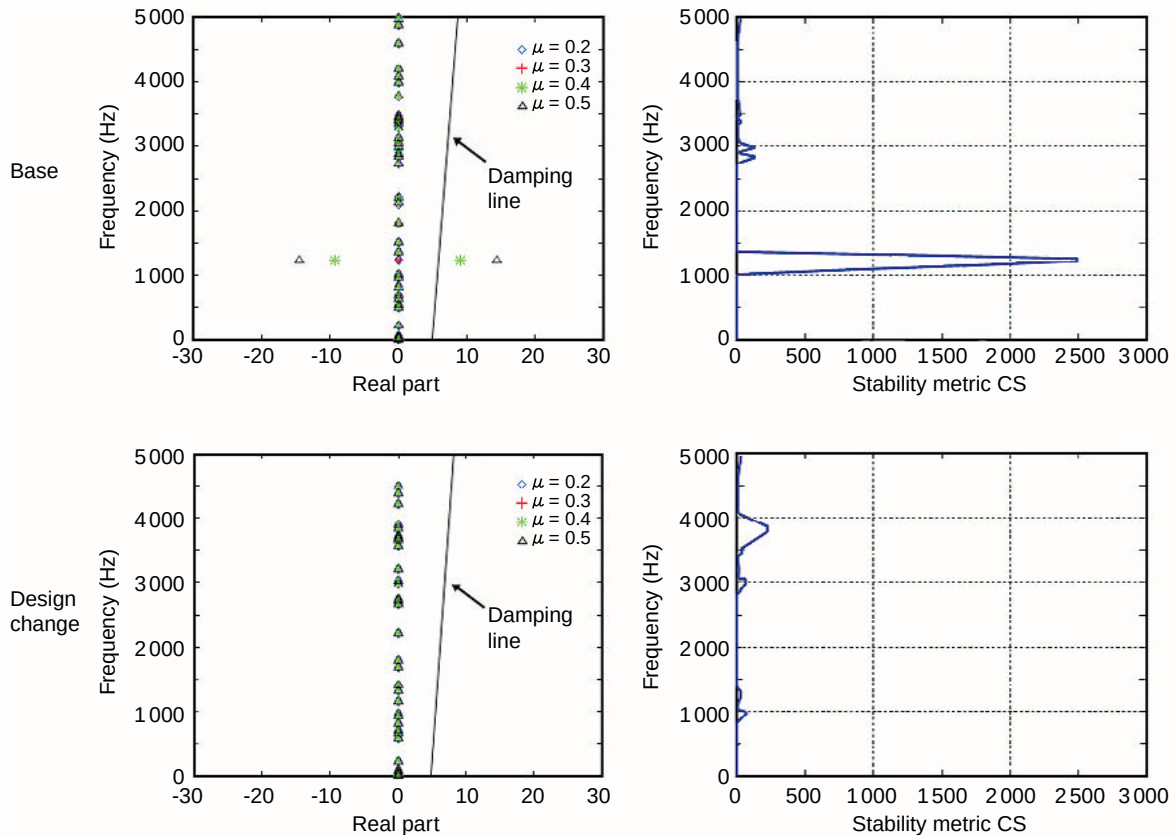


Fig. 14 Stability analysis of base specifications and specifications of design change

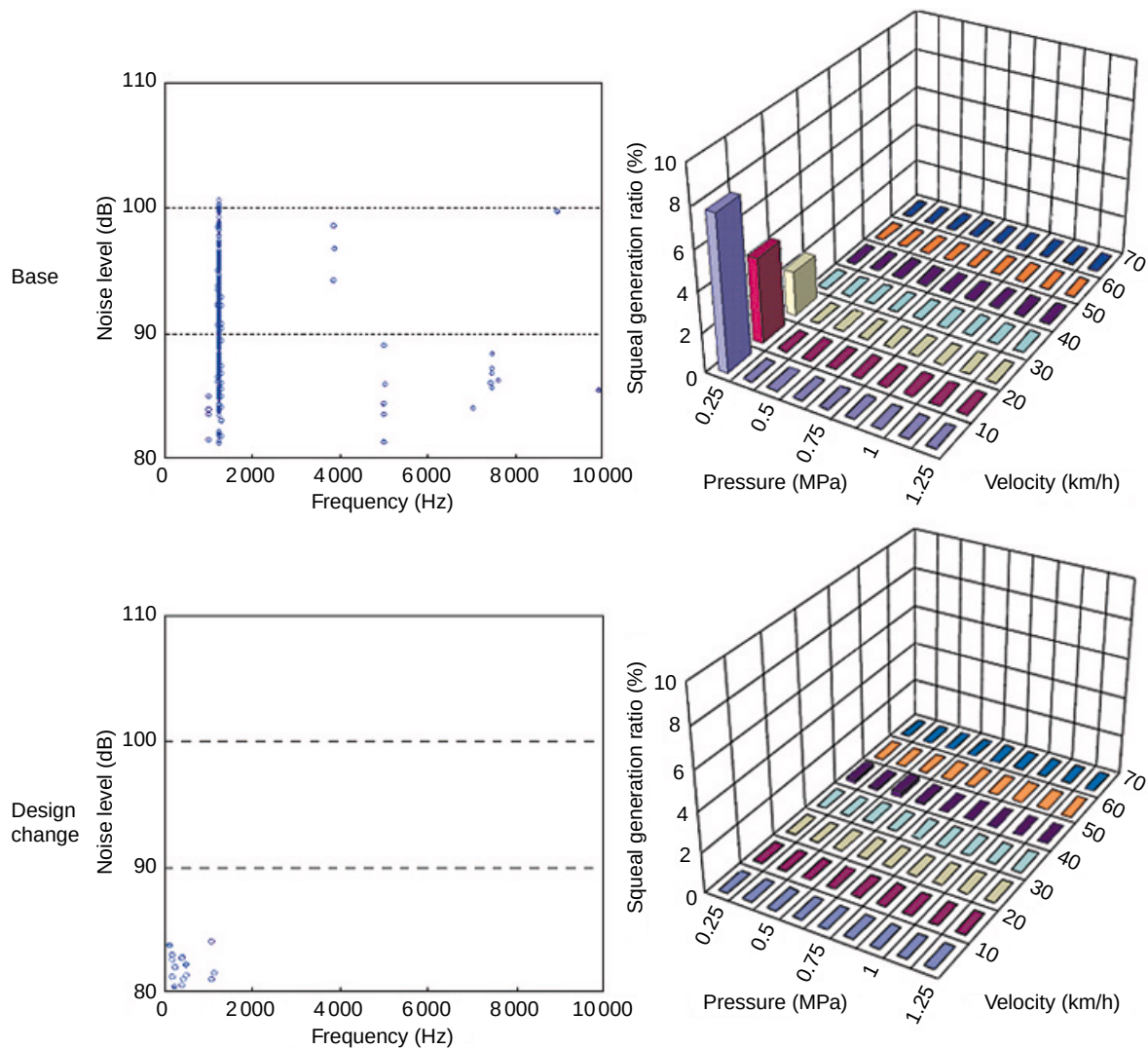


Fig. 16 Result of brake squeal noise tester matrix mode in base specifications and specifications of design change

7. Conclusion

A method to address the phenomenon of low frequency brake squeal noises in motorcycles easily and in a short period of time has been developed:

- (1) The equation was made simple by converting the complicated equation of motion for the system from a physical domain to a modal domain. The method of calculating a complex eigenvalue analysis based on the normal mode analysis results in a short period of time was established.
- (2) A new evaluation scale was defined for the instability mode wherein brake squeal occurs due to the coupling of two vibration modes in the brake system. This scale uses coupling strength as the basis for evaluation.
- (3) A method for setting the boundary conditions, which expresses the squeal noise occurrence state of a brake system, was established.
- (4) By revising the configuration of areas with high rates of strain energy contribution in the brake squeal generating mode, the countermeasures for the issue could be effectively applied.

Based on the above, we have identified the modes

affecting squeal and clarified the countermeasure methods. Further, we verified the effectiveness of this approach in predicting squeal noise in the design stage.

References

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